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(54) APPARATUS FOR CRUSHING ROCK MATERIAL AND THE LIKE, UTILIZING COMPLEX SONIC WAVE ACTION

(71) I, ALBERT GEORGE BODINE, a Citizen of the United States of America, of 7877 Woodley Avenue, Van Nuys, California, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to me, and the method by which it is to be performed to be particularly described in and by the following statement:—

This invention relates to apparatus for crushing rock material and the like utilizing complex sonic wave action and more particularly to such apparatus in which sonic energy is used both to crush the material and to efficiently feed such material through the crushing jaws.

In my British Patent Specification No. 1,153,076 an apparatus is described in which sonic energy is applied to one or both of a pair of crushing jaws which form part of a resonant vibration circuit. Rock material to be crushed is passed between the jaws through a wedge-shaped passage formed therebetween and acted upon by such energy so that such rock material is efficiently crushed to a particle size determined by the dimensions of the exit end of the passage. In the said apparatus the sonic energy is generated along a single axis so as to cause solely longitudinal jaw vibration, this in interests of obtaining maximum utilization of the sonic energy for the crushing operation. It has now been found that while the use of such a single-axis vibratory mode provides highly efficient crushing action, certain difficulties are encountered in some situations in properly passing the rock material through the wedge-shaped jaws, in some cases the rock material popping up out of the jaws. Further, in depending solely upon gravity feed, the taper angle of the feed passage must be limited in the interests of efficient feed-through, thus placing a distinct limitation on the particle size reduction capabilities of the crushing apparatus.

The apparatus of this invention, by utilizing a complex force wave pattern having a downward thrusting force component, enables a substantial increase in the throughput of the material being crushed and makes possible the utilization of a substantially wider angle between the crushing jaws thereby increasing the rock particle size reduction capabilities of the apparatus. In addition, the problems encountered with rock material being forced upwardly out from between the crushing jaws in the crushing process are substantially eliminated by virtue of the downward force components.

The improvement is achieved in the apparatus of this invention by positioning, operating and designing the force generation system to produce a complex alternating force output including both longitudinal and transverse force components. In certain embodiments of the device of the invention complex wave motion having relatively large-amplitude longitudinal components as compared with the transverse components are utilized while in other embodiments, apparatus for generating forces having transverse components of the same order of the longitudinal components thereof are described, the particular type of complex wave action utilized depending upon the particular application requirements involved. In certain embodiments of the device of the invention, the orbiting mass oscillator units which are utilized to generate the sonic energy are mounted right in the jaws themselves to enable the efficient driving of such oscillator units with a short shaft from the single drive motor.

Specific embodiments of this invention will now be described, by way of example, and referring to the accompanying drawings, wherein:

Fig. 1 is a top plan view of a first embodiment of the device of the invention;

Fig. 2 is a side elevational view in cross-section of the embodiment shown in Fig. 1;

Fig. 3 is a side elevational view in cross-section of an oscillator unit that may be uti-

lized in the embodiment of Figs. 1 and 2;

Fig. 4 is a side elevational view in cross-section of an oscillator unit that may be utilized in lieu of the dual oscillator shown in connection with the first embodiment of the device of the invention;

Fig. 5 is a drawing illustrating the basic structural components of a second embodiment of the device of the invention;

Fig. 6 is a drawing illustrating the basic structural components of a third embodiment of the device of the invention;

Fig. 7 is a top plan view with partial cut-away section of a fourth embodiment of the device of the invention;

Fig. 8 is a cross-sectional view taken along a plane indicated by 8—8 of Fig. 7; and

Fig. 9 is a cross-sectional view in elevation illustrating a replaceable jaw device that may be utilized in conjunction with the embodiment of Figs. 7 and 8.

It has been found that most helpful in analyzing the operation of the device of this invention to analogize the acoustically vibrating circuit involved to an equivalent electrical circuit. This sort of approach to analysis is well known to those skilled in the art and is described, for example, in Chapter 2, of Sonics, by Huter and Bolt, published in 1955 by John Wiley and Sons. In making such an analogy, force, F, is equated with electrical voltage, E; velocity of vibration, u, is equated with electrical current, i; mechanical compliance, C_m, is equated with electrical capacitance, C_e; mass, M, is equated with electrical inductance, L; mechanical resistance (friction), R_m, is equated with electrical resistance, R; and mechanical impedance, Z_m, is equated with electrical impedance, Z_e.

Thus, it can be shown that if a member is elastically vibrated by means of an acoustical sinusoidal force, $F_o \sin \omega t$, (ω being equal to 2π times the frequency of vibration), that

$$Z_m = R_m + j(\omega M - \frac{1}{\omega C_m}) = \frac{F_o \sin \omega t}{u} \quad (1)$$

Where ωM is equal to $\frac{1}{\omega C_m}$, a resonant condition exists, and the effective mechanical impedance, Z_m, is equal to the mechanical resistance, R_m, the reactive impedance components ωM and $\frac{1}{\omega C_m}$ cancelling each other out. Under such a resonant condition, velocity of vibration, u, is at a maximum, power factor is unity, and energy is most efficiently delivered to a load to which the resonant system may be coupled.

It is important to note the significance of the attainment of high acoustical "Q" in the resonant system being driven, to increase the efficiency of the vibration thereof and to provide a maximum amount of energy for the grinding operation. As for an equivalent electrical circuit, the "Q" of an acoustical vibration circuit is defined as the sharpness of resonance thereof and is indicative of the ratio of the energy stored in each vibration cycle to the energy used in each such cycle. "Q" is mathematically equated to the ratio between $\omega_o M$ and R_m($\omega_o = 2\pi f_o$, in which f_o is the resonance frequency of vibration. Thus, the effective "Q" of the vibrating circuit can be maximized to make for highly efficient high amplitude vibration by minimizing the effect of friction in the circuit and/or maximizing the effect of mass in such circuit.

Of significance in the implementation of the method and devices of this invention, is the high acceleration of the components of the elastic resonant system that can be achieved at sonic frequencies. The acceleration of a vibrating mass is a function of the square of the frequency of the drive signal times the amplitude of vibration. This can be shown as follows:

The instantaneous displacement, y, of a sinusoidally vibrating mass can be represented by the following equation:

$$y = Y \cos \omega t \quad (2)$$

where Y is the maximum displacement in the vibration cycle and ω is equal to $2\pi f$, f being the frequency of vibration.

The acceleration, a, of the mass can be obtained by differentiating equation (2) twice, as follows:

$$a = \frac{d^2y}{dt^2} = -Y\omega^2 \cos(\omega t) \quad (3)$$

The acceleration, a, thus is a function of $Y(2\pi f)^2$. At resonance, Y is at a maximum and thus even at moderately high sonic frequencies, very high accelerations are achieved making for correspondingly high vibrational forces at the grinding interfaces.

In considering the significance of the parameters described in connection with equation (1), it should be kept in mind that the total effective resistance, mass, and compliance in the acoustical vibration circuit are represented in the equation and that these parameters may be distributed throughout the system rather than being lumped in any one component or portion thereof.

It is also to be noted that an orbiting mass oscillator may be utilized in the device of the invention that automatically adjusts its output frequency to maintain resonance with changes in the characteristics of the load. Thus,

in the face of changes in the effective mass and compliance presented by the load, the system automatically is maintained in optimum resonant operation by virtue of the "lock in" characteristics of applicant's unique orbiting mass oscillator. The orbiting mass oscillator automatically changes not only its frequency but its phase angle and therefore its power factor with changes in the resistive impedance load to assure optimum efficiency of operation at all times.

Referring now to Figs. 1 and 2, a first embodiment of the device of the invention is illustrated. This first embodiment is structurally the same as one of the embodiments described in the aforementioned specification except for the phasing of the oscillator rotors. Stationary jaw member 14 which provides a high inertia mass is supported on I-beam support 13 which in turn is mounted on base member 10. Jaw member 14 has the inertia necessary to withstand or absorb a large periodic force impulse in the operation of the crushes without substantial yield or vibration. Mounted opposite stationary jaw member 14 is vibratory jaw member 15 which is in the general form of a rectangular block. Jaw member 15 is supported on I-beam 24 which in turn is supported on base member 10. Coupled to vibratory jaw member 15 by means of coupler flange 17 is elastic shaft member 16 which is preferably made of an elastic steel. Shaft member 16 is supported on I-beam support 23 which is mounted on base 10 by means of split mounting block 22. Side straps 23a are fastened at opposite ends thereof to block 22 and stationary jaw 14 to steady these members.

Attached to the end of elastic vibratory shaft 16 is orbiting mass oscillator 20. Orbiting mass oscillator 20 comprises a casing 35 having a pair of rotors 63 and 64 mounted for rotation in races formed thereon. Rotors 63 and 64 are driven in opposite directions at the same rotation speed by means of motor 71 through gear train 66. Motor 71 is supported on stand 70 which is mounted on base 10.

Referring now additionally to Fig. 3, rotors 63 and 64 are phased with respect to each other so that one of the rotors is slightly ahead of the other in its rotation cycle. Thus, when, for example, rotor 64 is in the "12 o'clock" position, rotor 63 may be in the "7 o'clock" position (instead of the "6 o'clock" position as in the aforementioned Patent Specification). By offsetting the phasal relationship of the rotor in this manner, a gyratory force is imparted to elastic vibratory shaft 16 causing it to vibrate in an elliptical path. The gyratory force pattern can be vectorially resolved into a larger force vector along the longitudinal axis of shaft 16 and a smaller force vector transverse to this longitudinal axis, such force vectors being indicated by arrows 40 and 41, respectively. The rotation speed of motor 71 is adjusted to cause resonant vibration of the vibration system including shaft 16 and vibratory jaw 15 to set up a standing wave pattern in shaft 16 as indicated by graph lines 42 with a velocity antinode indicated by "V" at the oscillator end of the shaft being transformed into a relatively low velocity indicated by "V" at the end of the shaft coupled to the high mass crushing jaw 15.

Sonic generator 20 may be of a type described in U.S. Patent Specification No. 3,217,551 which is illustrated schematically in Fig. 3. Briefly, this oscillator comprises a pair of rotors 63 and 64 which are mounted in housing 35 for rotation about separate races 38 and 39 formed by the inner walls of the housing. The rotors have gear rings 45 and 46 attached thereto which matingly engage gear rings 47 and 48, respectively, formed in the housing. Rotors 63 and 64 are driven in opposite directions by means of motor 71 which drives gear train 66 which in turn is coupled to shafts 53 and 54 through universal joints 59 and 56 respectively. Shafts 53 and 54 are coupled to rotors 64 and 63, respectively, through universal joints 58 and 55. As already noted, rotors 63 and 64 are phasally arranged with respect to each other so that while a large part of the transverse vibrational components effectively cancel each other out, leaving a predominantly longitudinal vibration mode in the output, there still remains a significant amount of transverse vibrational component as indicated by arrows 41, the importance of which will now be explained in connection with Fig. 2.

Rock material 32 to be crushed is fed into hopper 26 from feeder 31. This material passes from the hopper into a wedge-shaped passage formed beween the sloped wall 14a of stationary jaw member 14 and the straight wall 15a of vibratory jaw member 15. Jaw member 15 resonantly vibrates towards and away from jaw member 14a in the directions indicated by arrows 40 with relatively high force. The vibratory energy is transferred to the rock material and the rock material is rapidly fatigued and shattered. At the same time as the force component 40 of the resonant vibrational energy is achieving the crushing action, the force component 41 acts to move the rock material downwardly through the jaws and is phased with a sort of "pawing" action. Thus the rock material is kept moving through the jaws in an efficient manner.

If a greater downward force component is desired as, for example, where greater throughput is required or greater reduction in the particle size of the rock material is to be achieved with jaws having a wider angle, a single oscillator unit may be utilized as shown in Fig. 4 in place of the dual oscillator unit of Fig. 3. The single rotor 64 is driven by

motor 71 through gear train 66 in same manner as for the oscillator unit illustrated in Fig. 3. In this instance, however, the vertical force component is practically equal to the horizontal force component. It is to be noted that while the output of the single oscillator unit is in the form of a circular gyratory force patern, that in view of the greater stiffness or mass handling ability of shaft 16 in its longitudinal dimension than its lateral dimension, greater amplitude longitudinal vibration occurs than transverse vibration resulting in an eliptical gyration of the shaft with greater amplitude in the longitudinal than the transverse mode.

Referring now to Fig. 5, a second embodiment of the device of the invention is illustrated. In this embodiment, two separate orbiting mass oscillators 80 and 81 are mounted at the ends of resonant bars 83 and 84 respectively. The opposite ends of resonant bars 83 and 84 are attached to jaw members 86 and 87 respectively. Resonant bars 83 and 84 are supported on associated support members 88 and 89 and are vibrationally isolated therefrom by means of isolator members 90 and 91. The rotor of oscillators 80 and 81 are rotated at the same speed in opposite directions by motor means (not shown) to produce resonant gyratory vibration of their associated bar members. This results in a cooperative substantially circular motion of each of jaw members 86 and 87 as indicated by arrows 92 and 93, respectively. Longitudinal resonant vibration is set up in each of the bar members as indicated by resonant pattern 96 while transverse standing wave vibration is set up therein as indicated by wave pattern 97. This embodiment has an advantage in that in view of the cooperative substantially circular motion of the jaws relative sliding action therebetween is avoided, thereby greatly reducing friction effects and wearing of the jaw faces. Also the cooperative downward motion of the jaws tends to give the rock material 99 a maximum downward impelling force thereby greatly increasing the pumping action. This increases the throughput and enables the use of a wider jaw angle thereby making possible a greater reduction in the size of the rock particles. Further, the tendency of the rock material 99 to pop out from between the jaws is lessened. In the embodiment of Fig. 5, the mass of the resonant bars 83 and 84 is relatively large as compared with that of jaw members 86 and 87. This tends to greatly increase the relative magnitude of the downward or lateral wave portion of the composite wave which, as already noted, provides distinct advantages where greater downward pumping force is required.

Referring now to Fig. 6, a third embodiment of the device of the invention is illustrated. This embodiment differs from the previous embodiment in that the oscillators 80 and 81 are mounted right in the jaw members 86 and 87 themselves and in that additional resonant bar members 101 and 102 are attached to the jaw members to enhance the downward resonant vibration component. Resonant bar members 83 and 84 as for the previous embodiment are supported on supports 88 and 89 and vibrationally isolated therefrom by vibration isolators 90 and 91. The additional resonant bar members 101 and 102 extend vertically from the tops of jaw members 86 and 87, respectively, and are welded thereto.

This embodiment has the advantage that in view of the fact that the rotors are relatively close to each other, it is relatively easy to drive them from a single motor with an arrangement of the type illustrated in Fig. 3. The use of separate resonant bars for each of the vertical and horizontal force components assures optimum force output and highly efficient vibration thereby providing maximum force output at the crushing jaws. The positioning of the oscillators right in the jaws themselves with the resonant bars extending outwardly from the jaws enables a mechanization with four separate resonant vibration systems utilizing only two oscillators which are driven by a single drive motor. It is to be noted that all four resonant bars are preferably designed to resonate longitudinally at the same frequency so that with the oscillators driven by a single motor, optimum resonant operation is achieved with a resultant in-phase vibration of the jaws. The simple resonant vibration of the bar members is indicated by graph lines 105.

Referring now to Figs. 7 and 8, a fourth embodiment of the device of the invention is shown. In this embodiment, the jaw members 86 and 87 take the form of elastic tubular members with the oscillator units 80 and 81 being mounted directly therein. Here again the oscillator units may be of the type shown in Fig. 3. The oscillator rotors are driven by means of motor 115 through gear train 116 which is coupled to oscillator drive shafts 117 and 118. The jaw members 86 and 87 are supported in housing 112 and are vibrationally isolated from the housing by means of isolator pads 110 and 111 which locate the jaw members by means of bolts 114. In this embodiment, the jaws are resonantly vibrated in a wave pattern comprising two quadrature related waves transverse to the longitudinal axis of such jaws. This results in a circular in phase motion of the jaw members. This provides a highly effective crushing action and also highly effective downward pumping action. Oscillator units 80 and 81 may be pressfitted into their associated jaw members.

Referring now to Fig. 9, a modification of the embodiment of Figs. 7 and 8 wherein removable jaws are attached to tubular members 86 and 87 is illustrated. Jaw members

119 and 120 are welded to the sides of tubular members 86 and 87 and are utilized to perform the crushing action rather than the sides of the tubular members. Otherwise the structure and operation are identical as described for the previous embodiment. This embodiment has the advantage in that the jaws can be removed and repaired and replaced as necessary, thus avoiding the necessity for replacing the elongated tube members themselves.

The apparatus of this invention, thus by providing a downward pumping action, in addition to the sideways crushing action, affords the advantages of greater throughput and a capability for greater reduction in rock size in a single operation than heretofore possible and avoids the popping of rock material out from between the jaws with the vibratory crushing action.

WHAT I CLAIM IS:—

1. A crushing apparatus comprising a pair of opposed crusher jaws, at least one of which is vibratory toward and away from the other; means for feeding material to be crushed between said jaws;
a vibration generator adapted to deliver a complex alternating output force; and
an elastically vibratory wave transmission system comprising a longitudinally extending member of solid elastic material intercoupled between said generator and said vibratory jaw so as to receive said alternating force, undergo corresponding elastic vibration and impart vibration to said vibratory jaw; and means for driving said generator to produce a complex alternating force output which vibrates said jaw tooth both in up and down directions and in a direction toward and away from the other jaw.

2. The apparatus according to claim 1 wherein said generator comprises a pair of orbiting mass oscillators and means for rotating said oscillators in opposite directions at the same speed, said oscillators being phasally offset with respect to each other to generate both longitudinal and transverse phased force components in said wave transmission system.

3. The apparatus according to claim 1 wherein both of said jaws are vibratory toward and away from each other and wherein said generator comprises separate oscillator means for each of said jaws, said elasticially vibratory wave system comprising a similar member of solid elastic material intercoupled between each of said jaws and each of said oscillator means respectively, and means for driving said oscillator means to cause vibration of said jaws in a substantially circular path.

4. A crushing apparatus comprising a pair of opposed crusher jaws, both of said crusher jaws being vibratory toward and away from each other, means for feeding material to be crushed between said crusher jaws,
a vibration generator mounted on each of said crusher jaws,
a separate elastic longitudinally extending member of solid elastic material coupled to each of said crusher jaws, and
means for driving said generators to cause complex vibration of said crusher jaws to produce cooperative vibration thereof both towards and away from each other and cooperatively in an upward and downward direction.

5. The crushing apparatus according to claim 4 including a pair of mutually perpendicular solid elastic members coupled to each of said crusher jaws, each of said elastic members forming a separate resonant vibration system with an associated crusher jaw to cause vibration of said crusher jaws in a substantially circular path.

6. The crushing apparatus according to claim 5 wherein one of the elastic members of each of said pairs thereof is oriented vertically and the other of said elastic members are oriented horizontally.

7. The crushing apparatus according to claim 4 wherein there are two tubular members positioned alongside each other and each tubular member forms both one of said crusher jaws and the elastic member associated therewith.

8. The crushing apparatus according to claim 7 wherein said generators comprise orbiting mass oscillators mounted within each of said tubular members.

9. The crushing apparatus according to claim 7 including removable jaw members attached to said tubular members.

10. The crushing apparatus according to any preceding claim comprising a generator or generators capable of driving the jaw or jaws in such a way that the vibration of the jaw or jaws causes the jaw or jaws to have a downward movement during at least part of the time during which the jaws are moving toward one another.

11. Crushing apparatus substantially as described hereinbefore and with reference to the embodiment of Figures 1 and 2, or of Figures 5 or of Figure 6, or of Figures 7 to 9.

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1232644 COMPLETE SPECIFICATION

4 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 1

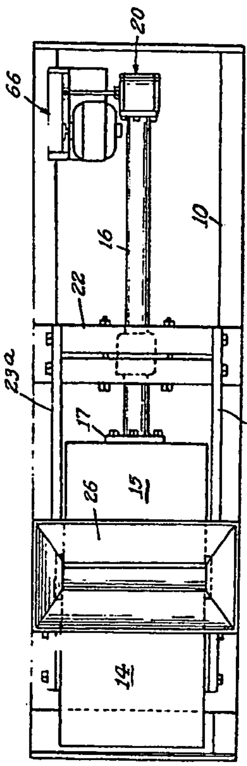

Fig.1

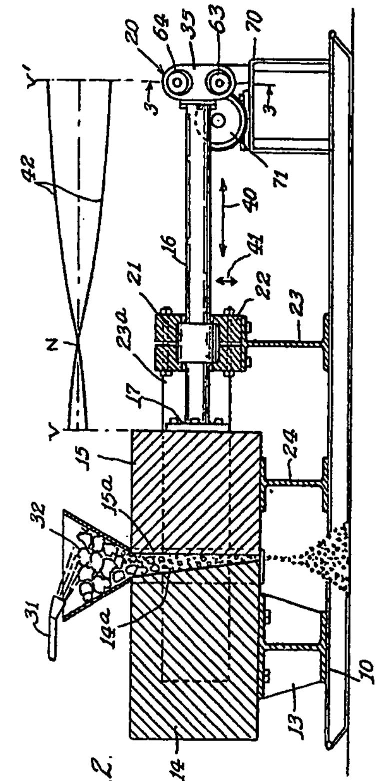

Fig.2

Fig.3

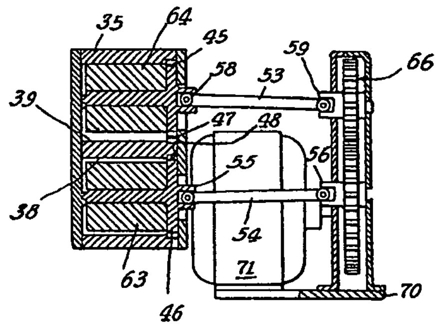
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Fig 4

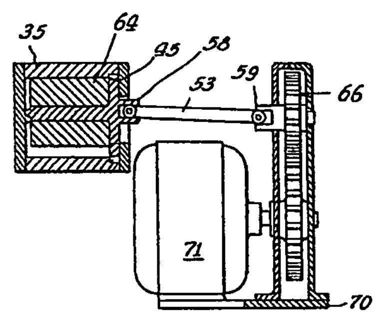
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Fig.5

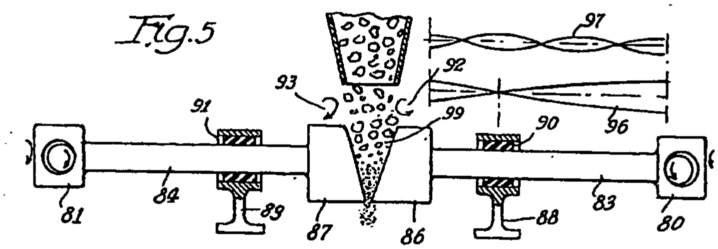
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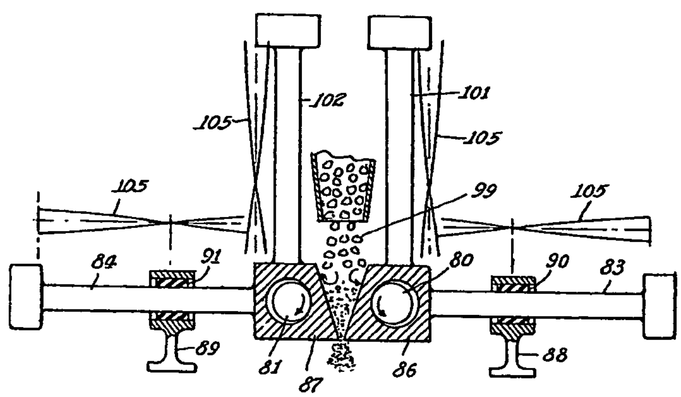
Fig.6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1232644 COMPLETE SPECIFICATION

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Sheet 4

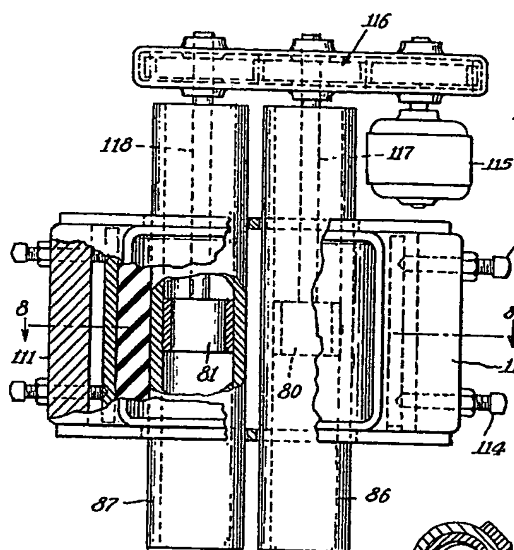

Fig.7.

Fig.9.

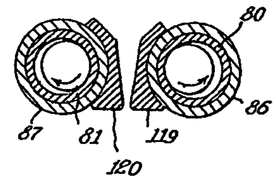

Fig.8